Conceptual Design of a Rubber Tracked Mini-Vehicle for Small Holders Using Off-Road Vehicle Engineering Techniques

A. F. Kheiralla, Yousif G. Alseed, Abbas Eltigani, Elhaj A. Yousif

Abstract — Knowledge and understanding of terramechanics and off-road vehicle engineering is becoming increasingly important making engineers better qualified to perform their jobs in agriculture, construction and military. Undergraduate engineering students should receive on-hands experience with these techniques to assist them in their future career. Furthermore, the significant feedback received from employers in industry stated that graduates need to be knowledgeable on terramechanics and off-road vehicle, and computer related aided methods. This paper is meant to supply engineering students with these techniques as a response to that feedback. A conceptual design of segmented rubber tracked mini-vehicle was successfully developed for small holders based on design formulas of these techniques. MathCAD and AutoCAD software were utilized in the designed of the vehicle. The vehicle was equipped with 12 kW gasoline engine and track size of 1000 mm² 200 mm. The overall dimensions of the vehicle were 2500 mm length, 1200 mm width, 1600 mm height and minimum ground clearance of 200 mm. The constructions of vehicle consisted of a chassis, power transmission system, steering system, and track system. The designed power transmission system consisted of clutch, differential unit, skid-steering and final drive with overall reduction ratio of 1:27. The designed skid steering system had steerability of 1.468 and power ratio of 1.125 for better mobility during turning. The vehicle was designed to give maximum drawbar power of 8 kW and maximum pulling ability of 3.63 kN with optimum slippage and tractive efficiency of 20% and 66.7%, respectively, under total weight of 9.81 kN.

Keywords — Off-road vehicle; Terramechanics; Segmented rubber; Skid-steering; Engineering Education

1. INTRODUCTION

WITH growing demands for improving mobility, high productivity and improve tractive performance of off-road vehicle such as wheeled and tracked type, over a wide range of terrains and in all seasons, have become increasingly popular in agriculture, military, earth moving, construction, mining and logging industry. To be competitive in today's global markets, product development must be on sound technological basis. This has led to a systematic study of vehicle terrain system. The study of overall performance of off-road vehicles in relation to its operating environment, and terrain is now become known as terramechanics.

Terramechanics has been primarily concerned with research of individual facets of vehicle-terrain interaction, such as the pressure-sinkage relationship and shear strength of the terrain, the mechanics of wheel-soil interaction and track-soil iteration, etc. Later it was recognized that for the benefits of vehicle designers and users, it is important to integrate the knowledge of various facets of terramechanics into a framework, which engineering practitioners can use as a tool to evaluate the overall design performance of off-road vehicles. Over the years, particularly since the World War II, a variety of mathematical models for predicting and evaluating off-road vehicle performance have been developed. They are range from entirely empirical to highly theoretical [1,2].

To retain the competitive edge in today's market, shortening the development and design cycle is of critical importance. Various computer-aided methods, such as computer simulation model for vehicle flexible tracks (NTVPM), computer simulation model for vehicles with rigid link tracks (RTVPM) and computer simulation model for off-road wheeled vehicles (NWVPM), have been developed to assist designers to optimize vehicle designs and procurement managers to select the most suitable vehicle candidate for a given mission and environment [3].

The computer simulation models NWVPM, NTVPM, and RTVPM have been successfully employed in assisting vehicle manufacturers in the development of new products and governmental agencies in the evaluation of vehicle candidates, in North America, Canada, Europe, Asia, and Africa. To demonstrate the applications of the comprehensive computer simulation models to evaluating the tractive performance of wheeled and tracked vehicles, an 8×8 wheeled vehicle, similar to a widely used light armoured vehicle (LAV), and a tracked vehicle with link tracks having relatively short-track pitch (referred to as flexible tracks earlier), similar to a widely used armoured personnel carrier (APC), are used as examples [4].

. Tracked vehicles have been recognized to give better tractive performances and cause less soil compaction than the wheeled vehicles. Steel tracked vehicles have been widely used in agriculture, earthmoving, and military, simply
because they are slow heavy, expensive and has high rates of wear in abrasive soils. However recent availability of new steel reinforcement and rubber moulding techniques have made it possible to construct rubber tracks of adequate strength and durability for use over a wide range of terrains and in tough environment. These newly developed rubber track are cheaper and lighter than steel tracks and allow the vehicles to operate on public roads.

Modifications on tire design have showed little potential for significance tractive improvements while rubber tracks have proved to have great potentials. Tractive performance test on rubber tracked vehicles claimed to develop more drawbar pull, higher tractive efficiency, and higher dynamic traction ratios with less slippage than wheeled vehicle in pneumatic tires. However, rolling resistance of tracked are higher than that of the tires, which resulted in the magnitudes of maximum tractive efficiencies being similar for both vehicles. On the other hand, because of better pull-push relationships for the tracked vehicles, the magnitude of pull and coefficient of traction at maximum tractive efficiency are much higher than the wheeled vehicle [4].

Knowledge and understanding of off-road-vehicle, terramechanics and computer related aided methods make engineers better qualified to perform their job in agriculture, construction and military. Undergraduate engineering students should receive on-hands experience with these techniques to assist them in their future career. Mastering these tools is anticipated to provide more job opportunities and good performances for engineering graduates. Furthermore, the significant feedback received from employers in industry stated that the engineering graduates need to be knowledgeable on off-road vehicle, terramechanics and computer related aided methods. Many text books on these techniques are currently available. Wong [5] a pioneer of terramechanics from Department of Mechanical and Aerospace Engineering, Carleton University, Ottawa, Canada had published his newly and up to date distinguished text book, Terramechanics and Off-road Vehicle Engineering: Terrain behaviour, Off-road Vehicle Performance and Design. Also Goering et al. [6] from the department of Agricultural and Biological Engineering, Illinois University, USA had published their famous text book, Off-road Road Vehicle Engineering Principles. This paper is meant to supply students with that technology as a response to that feedback. Before starting this proposed initiative project, the Faculty of Engineering, University of Khartoum did not offer any undergraduate course dedicated to off-road-vehicle engineering. The skills offered by such a project are needed in industry. However, there are no similar courses that were offered through other faculties at any of the universities in Sudan, while the existing other related courses are generalized and do not cover many important topics on off-road-vehicle engineering that suits its own need.

The main objective of this research work is to develop a conceptual design of segmented rubber tracked mini-vehicle for difficult terrain based on off-road vehicle engineering techniques using AutoCAD and MathCAD computer software. The work involves:
1. To design main chassis and power transmission systems
2. To design skid steering system,
3. To design rubber track system

II. MATERIALS AND METHODS

AutoCAD 2006 was used to develop the 3D conceptual design of the segmented rubber tracked mini-vehicle. The complete computation methods were pre-programmed in MathCAD 2000 professional to compute the vehicle various design parameters.

Wong collections, text books such as Terramechanics and Off-road Vehicle Engineering, Theory of Ground Vehicle, and Off-road Vehicle Engineering Principles were used to design vehicle elements (rubber track, steering system, track elements). The following criteria were considered in vehicle design:
1. Simplicity in design, ease of fabrication from locally available material,
2. Low cost within the reach of purchase in Sudan,
3. Two levers mechanism will ease vehicle operation,
4. Rubber will give better tracked floatation and reduces vehicle sinkage,
5. Rear larger diameter sprockets will reduce track slippage and increase vehicle traction,
6. Single and flexible main chassis with a wide open space, mounting points to accommodate transmission and track system, and various machine attachments,
7. Larger wheel base and shorter roller interval with track system will present a stable and smooth riding platform,
8. Robust in construction with improve comfort and visibility.

The proposed vehicle overall design consisted of power transmission system, track system, main chassis, steering system and vehicle body. The following assumptions are made in the computations used in the design of the proposed vehicle:
1. A total engine power for rubber tracked mini-vehicle is selected to be 16 hp (12 kW) at a rate engine speed of 3600 rpm, based on engine used at annual Quarter Scale Tractor for ASAE International Students Design Competition, East Moline, Illinois [7].
2. Vehicle theoretical speed is considered to be 10 km/h based on various off-road operations.
3. Vehicle total weight is considered to be 1 metric tone,
4. Vehicle's critical sinkage is consider to be 0.1 m,
5. Road wheel spacing is considered to be 0.225m to ensure good drawbar performance.
6. Vehicle centre of gravity is located at the midpoint of the track.

Fig. 1 depicted the overall configuration of the proposed mini rubber tracked vehicle.
\section*{A. Power Transmissions System}

The power transmission system is designed to transmit the engine power to the track system with minimum power loss. The proposed vehicle power transmission system consisted of multi-disk clutch, gearbox, differential unit, skid steering (clutch brake) system, and final drive sprocket gear. Transmit power is designed to account minimum losses and velocity with maximum reduction because vehicle needs low operation speed to improve traction (see Fig. 2).

\textbf{Overall Reduction Ratio:} The overall vehicle power transmission reduction was calculated based on engine speed and sprocket speed as follows:

\[ RR = \frac{Ne}{N_{spr}} \]  

Having engine speed of 3600 rpm and sprocket speed of 130 rpm based on theoretical travel speed of 10 km/h and sprocket diameter of 0.4 m, the computed value of reduction ratio is 1:27.6.

The overall reduction included gearbox, differential unit reduction ratio, and final drive sprocket gears reduction ratio by the following formula:

\[ RR = R_G \times R_D \times R_{FDG} \]  

Based on selected gearbox reduction of 1:2.3, the differential reduction ratio of 1.3 and final drive sprocket gear reduction ratio of 1:4, the computed overall reduction ratio is 1:27.1.

\textbf{Gearbox Unit:} The selected gearbox was a light duty, industrial dumper gearbox having three forward speeds and one reverse speed. The maximum actual gear ratio of the gear box was 1:10 while the minimum reduction was 1:2.3. This gear box unit was selected because it was ease in use and available in the market.

\textbf{Differential Unit Design:} Differential unit having two bevel gears were designed to transmit engine power from gearbox to skid steering shaft. The module of the bevel gears was computed by the following formula:

\[ T_m = \frac{2 \times T_p}{x \times b \times \pi \times N} \left( \frac{L}{L-b} \right) \]  

Knowing the rear shaft torque of 168 N.m, allowable stress of 593 MPa, tooth factor of 0.13, equivalent number of teeth of weaker pinion and gear, respectively, 17 and 50, pinion cone distance of 100 mm, face width of 0.033 m, the computed value of module was 5 mm. Furthermore, the pitch diameter of gear was calculated by the following formula:

\[ D = m \times N \]  

Having tooth module of 0.005 m, and gear and pinion teeth numbers of 50 and 17, the computed value of designed diameters for gear and pinion were 225 mm and 75 mm, respectively. Furthermore, the width of the designed gear was calculated by the following formula:

\[ b_g = k \times \pi \times m \]  

Having service factor of 4 and gear module of 5 mm, the computed value of gear width was estimated to be 63 mm.
Final Drive Gears Design: The design of the final derive sprocket was based on two spur gears and sprocket connected by shaft. Each final drive was designed to transmit final power from skid steering shaft to traction system. The final drive module is computed as follows:

\[ m = \frac{2 \times T_{spr}}{s \times \pi^2 \times k \times y \times Np} \quad (6) \]

Considering rear shaft torque of 864 N.m, allowable stress of 593 MPa, tooth factor of 0.098, tooth number of 18, service factor of 4 the computed value of module was 4 mm. Knowing the tooth number of 72 teeth, module of 0.004 m the computed value of designed gear was 28 cm and having service factor of 4, the computed value of gear width was estimated to be 45 mm.

Final Shaft Design: Final shaft diameter on final drive was designed based on applied torque and bending moment of under static loading condition as described below:

\[ d = \left( \frac{32 \times f_t}{s \times s_0} \right) \left( \frac{S_Y \times B_m}{5 \times E} \right)^{0.5} \left( 0.75 \times T_{spr} \times f \right) \quad (7) \]

Having bending moment of 2650 N.m, endurance strength of 283 MPa, sprocket torque of 881.53 N.m, yield tensile strength of 0.593 MPa, safety factor of 3, shaft length of 0.27 m, the computed shaft diameter was 0.045 m.

B. Steering System

In order to make a turn on tracked vehicle, it is necessary to drive one track faster than the other, causing the vehicle to turn toward the slower track. This called "skid steering" or "differential steering". In addition to steering the vehicle, a steering transmission system must be easy to use over rough and unfamiliar ground. Fig. 3 shows illustration of the proposed a skid steering attached to the final drive for the mini vehicle.

Clutch-Brake Steering: The mechanics of skid steering is described by Bekker [8] as shown in Fig. 4. At low speed, the centrifugal force may be neglected and the behaviour of motion of the tracked vehicle can be described by the following Eq.:

\[ m \cdot \frac{d^2 s}{dt^2} = F_o - F_t - R_{tot} \quad (8) \]

\[ I_s \cdot \frac{d^2 \theta}{dt^2} = B \left( F_o - F_t \right) - M_f \quad (9) \]

By solving the above motion differential equations, the thrust of the outside and inside tracks required to achieve a steady state turn can be computed by the following expressions:

\[ F_o = \frac{f_r \times W}{2} + \frac{M_f}{B} \quad (10) \]

\[ F_t = \frac{f_r \times W}{2} - \frac{M_f}{B} \quad (11) \]

Turning resistance moment can be determined as follows:

\[ M_f = \frac{\mu_t \times W \times L}{4} \quad (12) \]

Having coefficient of motion resistance of 0.1, tread vehicle of 1 m, coefficient of lateral resistance of 0.5, contact length of 1 m, the computed value of turning resistance moment was 1.226 kNm. Furthermore, the computed values of outside thrust and inside thrust were 1.717 kN and 0.74 kN, respectively. Brake force of 1.717 kN should be applied to inside track during turning.

Consequently, steerability of tracked vehicle can be determined by the following formula:

\[ \frac{L}{B} = \frac{2}{\mu_r} \left( \frac{C}{p} + \tan \phi - f_r \right) \quad (13) \]

Having contact length of 1 m, cohesiveness of 1.36 kN/m², contact pressure of 51.63 kPa, peat internal frictional angle 23.78° the computed value of steerability was 1.468. Having ratio of contact length to tread vehicle (L/B) equal 1 which is less than the computed steerability value of 1.468, this indicates that the vehicle will be able to steer on the specified terrain without spinning.

Since the brake of the inside track is fully applied, there will be no power loss in the brake. The power ratio \( P_t / P_m \) for a tracked vehicle can be calculating as follows:
Assume that operation terrain with coefficient of lateral resistance of 0.5, coefficient of motion resistance of 0.1 the computed value of power ratio was 1.125. This indicates that considerably more power is required during turn maneuver as compared to that in straight line motion.

C. Chassis and Track Assembly

The chassis and track configuration of the proposed vehicle are shown in Fig. 5. The overall dimensions of the designed chassis were 1800 mm length and 600 mm width. The chassis was made up of U shape mild steel of 100 × 50 ×5 mm size. The idler is mounted on swing arm. The sprocket is mounted to the final drive shaft, the road wheel assemblies consist of connecting rod moving vertically with helical springs, three road wheels are attached in each track and there are two sporting rollers connected to the chassis with swim arm.

The track design must enable the desired vehicle to move over unprepared terrain with uniform pressure distribution. The parameters for track system including track width, tack pitch, grouser height, idler diameter and location, sprocket diameters and location, road wheel diameter and road wheel spacing and ratio of road wheel spacing to track pitch. The proposed track system is shown in Fig. 6.

**Track Width:** The length of the track in contact with the ground and the level of pressure within the ground are the most important factors that influenced tracked vehicle performance. To evaluate the effects of track system configuration on vehicle ground pressure distribution and surface mat stiffness, it is important to study track ground contact length and width. The pressure-sinkage relationship for tracked vehicle on peat terrain up to the tracked critical vertical sinkage can be predicted using the following Eq. [2].

\[ p = k_p Z_c + 4 m_w Z_c^2 / D_b \]  

with  
\[ D_b = 4(L \times b) / (2(L + b)) \]  

The pressure can be expressed in terms of vehicle weight and track length and width as follows:

\[ P = W / (2 \times L \times b) \]  

Substituting pressure in Eq. 15, the track width can be determined by the following Eq.:

\[ b = W - (4 \times m_w Z_c^2 \times L) / (2Z_c(L \times k_p + 2 \times m_w Z_c)) \]  

Having total weight of 9.81 kN (1000 kg or 1 ton), total ground length of 1 m, critical sinkage is considered to be

\[ P_r = 0.5 \times \left(1 + \frac{\mu \times I}{2 \times f \times B}\right) \] (14)

**Track Pitch:** Short track pitch are commonly used with link track or belt track, in high speed tracked vehicles such as military fighting and logistics vehicles and off-road transporting vehicles. Track pitch was determined by following Eq.:

\[ Tp = Rs / 2.25 \] (18)

Considering road wheel spacing of 0.225 m, the computed track pitch was 0.1m.

**Track Grouser Size:** The use of grouser on tracks would be required to fully utilize the shear strength of the peat surface mat for generating tractive effort. The surface mat thickness of peat terrain was found to be about 0.1 m. In order to fully utilize the shear strength of the surface mat and to increase the traffic ability of the terrain the grouser height of the track is considered to be 0.06m [3].

**Road Wheel Diameter:** For high-speed tracked vehicles to minimize the vibration of the vehicle and of the track, relatively large diameter road wheels with considerable...
suspension travel. Road wheel diameter was determined by using the following Eq.:  
\[ D_r = (R_s - G) \]  
(19)

The gab is assumed to be 0.05 m, the computed road wheel diameter was 0.22 m.

Number of Road Wheel: Number of road wheel was determined by using the following Eq.:  
\[ \frac{L}{nr} = \frac{D_{rs} + D_{fi}}{2} \]  
(20)

Having contact length of 1 m, rear sprocket diameter of 0.4m, front idler diameter of 0.4m, road wheel diameter of 0.22 m, gab between consecutive spacing 0.05 m the computed road wheel number was estimated to be 3.

Sprocket Size and Location: The location of drive sprocket has a noticeable effect on the vehicle tractive performance. Wong et al. [9] reported that in forward motion, the top run of the track is subjected to higher tension when the sprocket is located at the front than when the sprocket is located at the rear. The size of the sprocket can be determined from the relationship between the sprocket torque and vehicle speed fluctuation as follows:  
\[ T_{spr} = \frac{kW \times 1000 \times 60}{2 \times \pi \times N} \]  
(21)

Having the engine power of 12 kW and sprocket revolution of 130 the computed value of sprocket torque was 800 Nm. The sprocket diameter was determined by using the following Eq.:  
\[ D_{spr} = \frac{T_p}{\sqrt{1 - (1 - \delta)^2}} \]  
(22)

Having track pitch of 100 mm, speed fluctuation of 3.15%, the computed value of sprocket diameter of 400 mm.

Idler Size and Location: The track entry angle is significantly affected the vehicle front idler size and tractive performance. Therefore, from the relationship between the vehicle sinkage, track entry angle can be identified. Both the vehicle track entry angle at front idler and sinkage decrease with increasing vehicle front idler diameter as shown in Fig. 7. Critical sinkage of the vehicle is considered to 100 mm, the corresponding front idler diameter and track entry angle were found 400 mm and 78°, respectively.

D. Tractive Performance

Tractive Effort: Tractive effort of the vehicle for peat terrain can be computed by the following Eq.:

\[ F_t = (2 \cdot A \cdot C + W \cdot \tan(\phi)) \left[ 1 - \frac{W}{r \cdot L} \right] \]  
(23)

The selected terrain having cohesiveness of 1.36 kN/m², peat internal frictional angle of 23.78º, total engine weight was 9.81 kN, slippage was 20%, slope angle was 25º, shear deformation modules was 1.19 cm, track entry angle was 78º, the computed value of tractive effort was 4.549 kN.

Motion Resistance: Motion resistance of the tracked vehicle can be splited in to internal and external resistance. The motion resistance due to terrain compaction was determined by the following Eq.  
\[ R_c = 2 \cdot b \cdot \left[ \frac{k_v \cdot Z_c^2}{2} + \frac{4}{3} \cdot \frac{m_w \cdot Z_c^3}{D_h} \right] \]  
(24)

Having hydraulic diameter of 0.08 m the computed value of motion resistance due to compaction was 598 N.

The motion resistance due to frictional losses of the vehicle moving components can be predicted using the following Eq.  
\[ R_{mf} = \frac{W}{10^6} (222 + 3 \cdot F_t) \]  
(25)

Theoretical velocity was considered to be 10 km/hr the computed value of motion resistance due to frictional losses was 252 N.

The motion resistance of the overall components of the track it can be determined by the following Eq.:  
\[ R_w = 2 \cdot b \cdot \left[ p_d \cdot Z_c^2 \cdot \tan \left( \frac{\phi}{2} \right) + C \cdot Z_c \cdot \tan \left( \frac{45 + \phi}{2} \right) \right] \]  
(26)

Having bulk density of the selected terrain of 1.53 kN/m³, the computed value of motion resistance for track elements was found to be 92.865 N.
The total motion resistance of the rubber tracked vehicle can be computed as the sum of the individual motion resistance components by:

\[ R_{t1} = R_c + R_m + R_n \]  

(27)

Based on the values of the individual motion resistance components, the total motion resistance is computed to be 923 N. Consequently the total motion resistance coefficient for the vehicle having pay load of 9.8 kN is computed to be 9.5%. Based on Wong [11], the total motion resistance for the vehicle of rigid link on soft terrain should be in the range of 6 - 9% of the total weight. From the computed result, it is found that the total motion resistance agrees with recommended total motion resistance of vehicle.

**Drawbar Pull and Power:** Drawbar pull and power is the ability of vehicle to push or pull over unprepared terrain and the power available on drawbar, the draw bar pull can be predicting by the following formula:

\[ Dp = F_i - R_{nt} \]  

(27)

The computed value of drawbar pull was 3.63 kN. The drawbar power is referred to as the potential productivity of the vehicle, that is, the rate which productive work may be done. It can be computed by using the following Eq.

\[ P_d = \left( \frac{1}{367.2} \right) \times (Dp \times V_e) \]  

(28)

Having actual speed of 8 km/hr, the computed value of drawbar power was 7.9 kW

**Tractive Efficiency:** Tractive efficiency is used to characterize the efficiency of track vehicle in transforming the engine power to the power available at the drawbar. It can be computed by using the following Eq.:

\[ \eta_d = \frac{P_d}{P_e} \times (100\%) \]  

(29)

Having engine power of 12 kW, drawbar power of 8 kW computed value of tractive efficiency was 66.7%.

### III. RESULTS AND DISCUSSION

**A. Vehicle Configurations**

Based on the results of the computation design, a mini rubber tracked vehicle was successfully designed. The detail specifications of the designed vehicle are summarized in Table 1. The 3D conceptual design of complete rubber track mini-vehicle is shown in Fig. 8. The overall design of the designed vehicle consisted of power transmission system, track system, main chassis, steering system and vehicle body. The vehicle was powered by naturally aspirated air-cooled, 4 strokes, 1 piston (Kohler, Model K 341S) engine that has a rated power of 16hp@3600. The engine has fuel tank of 4.5 L capacity. The overall dimension of the designed vehicle was 2500 mm length, 1400 mm width, 1600 mm height, and minimum ground clearance of 200 mm. The designed vehicle had minimum clearance of 200 mm under gross static weight of one metric ton located at the mid point of contact area.

**Selected Terrain Parameters:** Peat terrain was selected as design condition for design of the vehicle. The overall selected peat terrain parameters are shown in Table 2. The selected area of peat was located in Sepang, Malaysia having 83.51% moisture content, 1.53 kN/m² peat bulk density, 1.36 kN/m² cohesive strength, 23.78 degree peat internal frictional angle, 1.19 cm shear deformation modules, 27.07 kN/m² mat strength, 224.38 kN/m² underlying peat stiffness [12].

**Chassis Configurations:** A flexible chassis for the vehicle was designed successfully. The overall dimensions of the designed chassis were 1800 mm length and 600 mm width. The chassis was made up of U shape mild steel of 100 × 50 ×5 mm size. The front idler is mounted on swing arm. The rear sprocket is mounted to the final drive shaft which is attached to the sprocket gear, the road wheel assemblies consist of connecting rod moving vertically with helical springs to allow vehicle to moving on obstacles, three road wheels are attached in each track and there are two sporting rollers are connected to the chassis with swim arm.

**Track Configuration:** A segment rubber track was designed successfully to move over selected peat terrain. Details configuration 3D conceptual design of chassis and track system is shown in Fig. 9. The designed track was 200 mm width, 3256 mm total length, 60 mm pitch, 60 mm grouser height, and 1000 mm track ground contact length. The track system configuration of 1 ton (9.81 kN) mini vehicle has three road wheels in each side with diameter of 220 mm, two sporting rollers with diameter of 150 mm in the top to ensure good tension, rear sprocket diameter of 400 mm, front idler diameter of 400mm, a location of center of gravity at the mid-point of the track for uniform pressure distribution.

**B Transmission System**

The transmission system was designed successfully to transmit the engine power to the rear axle with minimum power loss. The designed vehicle transmission system consisted of gearbox differential unit, disk clutch, power shaft, clutch brake skid-steering system, sprocket gears, the vehicle had overall gear reduction ratio of 1:27.

The bevel gears (differential unit) were designed to transmit power from gearbox to rear axle with 1:3 reduction ratio, each designed gear of 5 mm modules, 33 mm, face width and 100 mm cone distance, pinion of 15 teeth, 75 mm diameter, gear of 45 teeth and 225 mm diameter. Two spur gears for the final drive are designed to ensure enough clearance of designed vehicle and to transmit power and
torque from the rear shaft to sprocket, the designed gear each of 4 mm of modules, 45 mm, width, 300 mm, gear diameter and pinion diameter of 75 mm.

C. Skid Steering System

Skid-steering (clutch brake system) was designed successfully to enable the vehicle to steer on selected terrain easily the designed steering system consist of bevel gears, frictional clutch, drum brake and shaft. The computed out and inners thrust were found to be 1.717 kN and 0.784 kN, respectively. Therefore the designed brake force of 1.717 kN should be applied to inside track during turn was with resistance moment of 1.226 kNm. The computed steerability of tracked was found to be 1.468 which indicates that the vehicle will be able to steer on the specified terrain without spinning. The computed power ratio was found to be 1.125 which indicates more power is required during turning.

**Gearbox Selection:** The selected gear box was light weight industrial dumper having three forward speed and one reverse speed with a minimum reduction ratio of 2.3 and maximum reduction ratio of 10. This gear box unit was selected because it was ease in use and available in the market.

**TABLE 1: THE RUBBER TRACK MIN-VEHICLE SPECIFICATIONS**

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<thead>
<tr>
<th>Engine</th>
<th>Specifications</th>
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<tbody>
<tr>
<td>Model</td>
<td>Kohler, Model K 341S</td>
</tr>
<tr>
<td>Type</td>
<td>12 kW@3600 rpm</td>
</tr>
<tr>
<td>No. of cylinder</td>
<td>4 cycle, gasoline, air cooled</td>
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<td>Bore and stroke</td>
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**TABLE 2: PEAT TERRAIN PARAMETERS (ATAUR ET AL. [12])**

<table>
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<tr>
<th>Parameter</th>
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<td>$\gamma$ (kN·m$^{-2}$)</td>
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<td>$C$ (kN·m$^{-2}$)</td>
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<td>$M_m$ (kN·m$^{-3}$)</td>
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**Transmission system**

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<td>Sprocket gear</td>
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<td>Clutch system</td>
<td>disk</td>
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<td>Brake system</td>
<td>drum</td>
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<tr>
<td>Steering system</td>
<td>clutch brake skid-steering system</td>
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</tbody>
</table>

**Fig. 8** The 3D conceptual design of complete rubber track mini-vehicle

**Fig. 9** The 3D conceptual design of chassis and track

**International Conference on Trends in Industrial and Mechanical Engineering (ICTIME'2012) March 24-25, 2012 Dubai**
Final Drive Shaft: The final drives consist of two shafts of 45 mm to transmit the torque and motion. The low carbon steel (AISI No. 1006) was selected to be the designed material of shaft due to easily machining, availability and sufficient strength for torsion and bending moment. Each shaft with 270 mm, length, 45 mm, diameter supporting with connecting with last gear base at both sides.

D Tractive Performance

For vehicle tractive performance, the selected terrain had cohesiveness of 1.36 kN/m², peat internal frictional angle of 23.78°, total engine weight was 9.81 kN, slippage was 20%, slope angle was 25°, shear deformation modules was 1.19 cm, track entry angle was 78°, the computed value of tractive effort was 13.06 kN and motion resistance was 92.865 N. The vehicle was designed to give a maximum pull of 3.63 kN under gross weight of 9.81 kN and slippage of 20%. The maximum efficiency was found to be 66.7% on peat terrain and could be increased if better terrain is used.

IV. CONCLUSIONS

The following conclusions were based on the obtained results:
1. A conceptual rubber tracked mini-vehicle was successfully developed for small holders. The complete vehicle is shown in Fig. 8 and it is technical specifications is shown in Table 1.
2. Several software have been used during the design stage of the vehicle. MathCAD 2002 professional was used to calculate various components and parameters in the transmission system, track system, skid steering system and tractive performance. AutoCAD 2006 was used to develop the 3D concept of the vehicle.
3. The transmission system was successfully designed. The designed transmission system consisted of gearbox, differential unit, skid steering system and final drive. The designed transmission system have overall reduction ratio of 1.271.
4. The traction system was successfully designed. The designed traction system consisted of segmented rubber track having size of 200 mm width and contact length of 1000 mm for the designed vehicle weight of 9.81 kN in order to get the desired pressure distribution to limit track sinkage of 10 cm.
5. The clutch/brake skid steering system was successfully designed. The computed value of steerability was found to be 1.468 and power ratio of 1.125 which provide vehicle with better mobility during turning.
6. The maximum tractive efficiency of the designed vehicle at optimum track slippage of 20% is expected to be 66.7% on selected peat terrain.

The following recommendations were made for underlying results:
1. Upon the completion of this designed work, funds should be located to develop and evaluate the rubber track mini-vehicle.
2. Terramechanics and off-road vehicle engineering course should be undertaken in the newly proposed agricultural and biological engineering and relevant curriculum.

REFERENCES


NOMENCLATURE

\[ \begin{align*}
N_c & \quad \text{engine speed, rpm} \\
N_{sp} & \quad \text{sprocket speed, rpm} \\
RR & \quad \text{reduction ratio, dimensionless} \\
RGB & \quad \text{gear box reduction ratio} \\
RDG & \quad \text{differential reduction ratio} \\
RFDG & \quad \text{final drive sprocket gear reduction ratio} \\
m & \quad \text{minimum module, mm} \\
T_b & \quad \text{shaft torque, N.m} \\
s & \quad \text{allowable stress, MPa} \\
y & \quad \text{tooth factor} \\
N_p & \quad \text{equivalent Number of teeth} \\
L & \quad \text{cone distance, mm} \\
width & \quad \text{face width, mm} \\
A & \quad \text{contact area, m}^2 \\
L & \quad \text{contact length, m} \\
\end{align*} \]
C  peat terrain cohesiveness, kN/m²
φ  peat internal frictional angle, degree,
P_t  power consumption of tracked vehicle during a turn, kW
P_{st}  power consumption of tracked vehicle during straight line motion, kW,
μ_s  coefficient of lateral resistance, dimensionless,
b_g  gear width, mm
s  allowable stress, MPa
k  service factor, dimensionless
y  tooth factor, dimensionless
f_s  safety factor, dimensionless
SY  yield tensile strength, MPa
BM  bending moment, N.m
SE  endurance strength, MPa
s  the displacement of the center of gravity of the vehicle, m
F_o  outside thrust, N
F_i  inside thrust, N
M_r  turning resistance moment, N.m
μ_r  coefficient of lateral resistance, dimensionless,
F_r  coefficient of motion resistance,
B  tread vehicle, m
W  Total weight, kN,
p  contact pressure, kPa
m_w  mat strength, kN/m³
k_p  peat strength, kN/m³
Z_c  critical sinkage, m
D_h  hydraulic diameter, m
T_p  track pitch, m
R_s  road wheel spacing, m
G  is the gab between consecutive road-wheel
n_r  road wheel number
L  contact length, m
D_{rs}  rear sprocket diameter, m
D_{fi}  front idler diameter, m
D_r  road wheel diameter, m
kW  engine power, kW
N  sprocket revolution, rpm
ST  sprocket torque, Nm
T_p  track pitch, m
δ  speed fluctuation, percentage
K_w  shear deformation modules
i  slippage, percentage
F_t  tractive effort
R_c  motion resistance due to compaction, kN
R_m  motion resistance for track elements
D_p  drawbar pull, kN
F  tractive effort, kN
R_s  total resistance, kN
V_a  actual velocity
P_d  drawbar power, kW
P_e  engine power, kW